## §2. Deformation of Coil Supporting Structure during Cooling Down

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Cryogenic components in LHD, superconducting coils and a coil supporting structure, are cooled down to the liquid helium temperature. The maximum diameter among the supporting structure which is made of 316-type stainless steel is 13 m. A thermal contraction between room temperature and liquid helium temperature is 0.3 % for the stainless steel. Coolant pipes were attached to the structure with a high thermal conductive resin. Fig. 1 shows 1/10 section of the supporting structure with the cooling paths. Spatial temperature distribution in the structure results in a mechanical stress generation. To investigate the stress distribution under an actual thermal condition, temperature and stress distribution of the coil supporting structure was calculated by using the finite element method considering a temperature dependent of material properties [1].

All components in LHD is a symmetric in every $\pi/5$ radian for its toroidal direction, and a shape of upper and lower part is identical. A half-upper section of toroidal angle $\pi/5$ was modeled and three dimensional shell elements were used. Total number of elements and nodes are 4479 and 4581, respectively. Physical properties needed were defined as a function of temperature except Poisson's ratio. There were two steps of calculation: spatial temperature distribution and structural calculation. These calculations were solved by using ANSYS5.4.

We used exponential function to simulate temperature gradient in the cooling pipes as,

$$T_l = T_{\text{inlet}} + \alpha\{1 - \exp(-\beta \frac{l}{l_0})\} \tag{1}$$

where $l_0$ is a length of a cooling pipe, $l$ is a distance from the inlet, $T_l$ is a temperature at l, $\alpha$ and $\beta$ are parameters that express a magnitude of temperature change. According to the experience of the second operation for LHD, the parameters in eq. (1) were assumed as

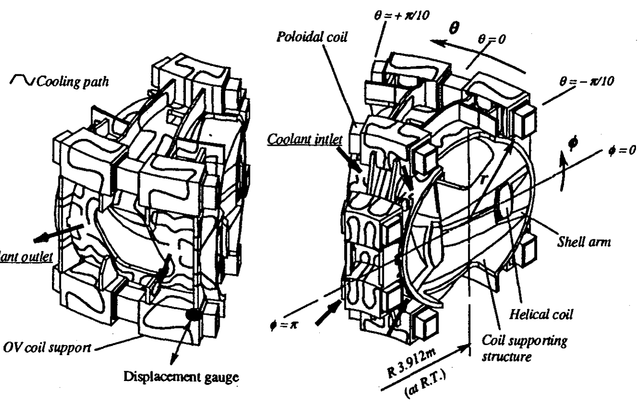


Fig. 1. Coolant paths and measured point of displacement for the coil supporting structure.

$$\alpha = \frac{9}{8}(T_{\text{outlet}} - T_{\text{inlet}}), \qquad \beta = \ln 9 \tag{2}$$

Temperature was set to the nearest node points to the actual cooling path. Transient temperature for every node was solved in this calculation. The temperature distributions obtained from the thermal analysis were set to all nodes at certain time. Boundary conditions for sector to sector were cyclic so that the displacement and rotation on each side should be equal. For upper to lower boundary, displacement and rotation should be odd-symmetry.

Fig. 2 shows the result of analysis for displacement described in fig. 1 against an averaged outlet temperature. The results between analysis and measurement showed a good correlation. Strain distribution was calculated in the analysis simultaneously. The maximum equivalent stress of 124 MPa appeared at the corner of the upper port area 4 days after starting cooling down, when the inlet and the outlet temperatures were 193 K and 237 K, respectively. Stress in other area was not exceeded 60 MPa. The strain of Inner-equator was the largest among them. The inner area of the component was cooled first but the other area was still warm and did not deform compare with inner area. Then the large tensile subjected to the inner area.

Measured strains from the actual component during operation seemed to have unknown error. The measured strain possibly includes apparent strain that depends on temperature and other kind of influence. We defined a strain difference between $\theta$ and $\phi$ directions. These two strains were obtained from one bi-axial strain gauge. Since the temperatures between each two axes could have been same, the apparent strain was canceled by the subtraction. As the result, the tendency and value of strain agreed with analytical one very well.

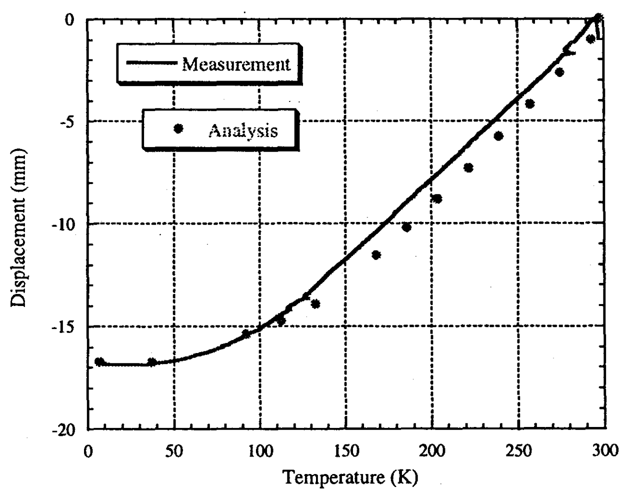


Fig. 2. Displacement at OV coil support during cooling down.

Reference
1) Tamura, H. et al., Proc. ICEC17, (1998) 871